

**Active volume dimensions**, from earlier analyses:

$$r_{Xe} = 0.53 \text{ m} \quad l_{Xe} = 1.3 \text{ m}$$

We consider using a field cage solid insulator/light tube of 3 cm total thk., and a copper liner of 12 cm thickness, including all tolerances and necessary gaps.

$$t_{fc} := 3 \text{ cm} \quad t_{Cu} := 12 \text{ cm}$$

**Pressure Vessel Inner Radius:**

$$R_{i\_pv} := r_{Xe} + t_{fc} + t_{Cu} \quad R_{i\_pv} = 0.68 \text{ m}$$

**Pressure Vessel inner Diameter:**

$$D := 2R_{i\_pv}$$

**Temperatures:**

For pressure operation, the temperature range will be 10C-30C. For vacuum operation, the temperature range will be 10C to 150C (bakeout).

**Vessel wall thicknesses**

**Material:**

We use 316Ti for vessel shells and flanges due to its known good radiopurity and strength.

**Design Rules: division 1**

316Ti is not an allowed material under section VIII, division 2, so we must use **division 1** rules. The saddle supports are however, designed using the methodology given in div. 2, as div. 1 does not provide design formulas (nonmandatory Appendix G)

Maximum allowable material stress, for sec. VIII, division 1 rules from ASME 2009 Pressure Vessel code, sec. II part D, table 1A:

$$S_{\max\_316Ti\_div1} := 20000 \text{ psi} \quad -20F - 100F$$

Youngs modulus

$$E_{SS\_aus} := 193 \text{ GPa}$$

**color scheme for this document**

input    check result (all conditions should be true (=1))

$$xx := 1 \quad xx > 0 = 1$$

Choose material, then maximum allowable strength is:

$$S := S_{\max\_316Ti\_div1}$$

Maximum Operating Pressure (MOP), gauge:

$$MOP_{pv} := (P_{MOPa} - 1 \text{ bar}) \quad MOP_{pv} = 14 \text{ bar}$$

Minimum Pressure, gauge:

$$P_{\min} = -1.5 \text{ bar} \quad \text{the extra 0.5 atm maintains an upgrade path to a water or scintillator tank}$$

Maximum allowable pressure, gauge (from LBNL Pressure Safety Manual, PUB3000)  
at a minimum, 15% over max operating pressure; this is design pressure at LBNL:

$$MAWP_{pv} := 1.1 MOP_{pv} \quad MAWP_{pv} = 15.4 \text{ bar}$$

$$P := MAWP_{pv}$$

Vessel wall thickness, for internal pressure is then (div 1), Assume all welds are type (1) as defined in

UW-12 , are fully radiographed so weld efficiency:

$$E := 1$$

<---note: it is not clear if butt welds using ceramic backing strips will qualify as type (1)

Minimum wall thickness is then:

$$t_{pv\_d1\_min\_ip} := \frac{P \cdot R_{i\_pv}}{S \cdot E - 0.6 \cdot P} \quad t_{pv\_d1\_min\_ip} = 7.75 \text{ mm}$$

We set wall thickness to be:

$$t_{pv} := 10 \text{ mm}$$

$$t_{pv} > t_{pv\_d1\_min\_ip} = 1$$

### Maximum Allowable External Pressure

#### ASME PV code Sec. VIII, Div. 1- UG-28 Thickness of Shells under External Pressure

Maximum length between flanges  $L_{ff} := 1.6 \text{ m}$

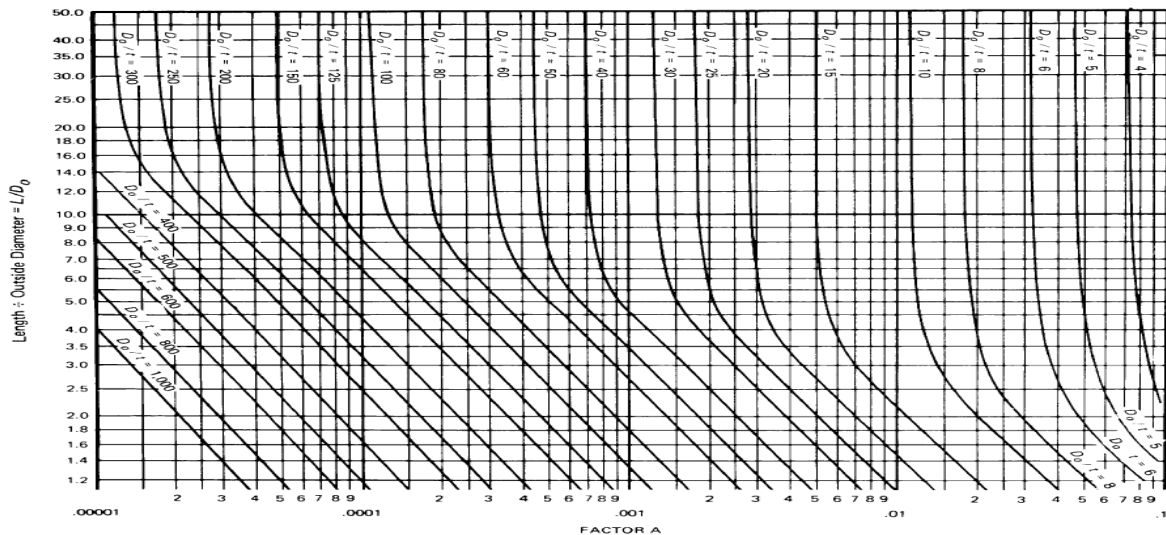
The maximum allowable working external pressure is determined by the following procedure:

Compute the following two dimensionless constants:

$$\frac{L_{ff}}{2R_{i\_pv}} = 1.2 \quad \frac{2R_{i\_pv}}{t_{pv}} = 136$$

From the above two quantities, we find, from fig. G in subpart 3 of Section II, the factor A:

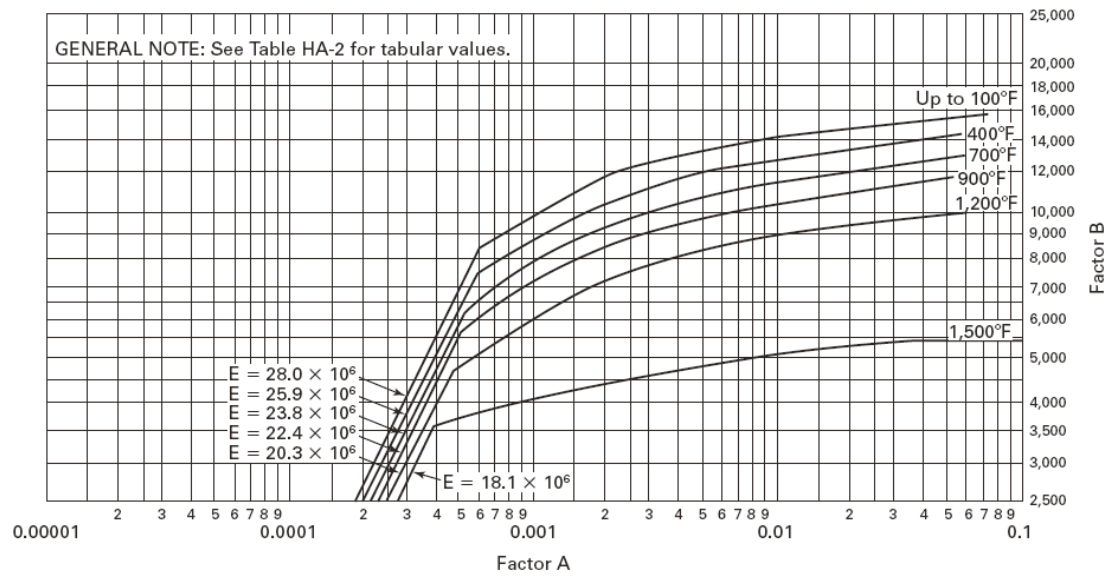
FIG. G GEOMETRIC CHART FOR COMPONENTS UNDER EXTERNAL OR COMPRESSIVE LOADINGS (FOR ALL MATERIALS) [NOTE (14)]



$$A := 0.0005$$

Using the factor A in chart (HA-2) in Subpart 3 of Section II, Part D, we find the factor B ( @ 400F, since we may bake while pulling vacuum):

FIG. HA-2 CHART FOR DETERMINING SHELL THICKNESS OF COMPONENTS UNDER EXTERNAL PRESSURE  
DEVELOPED FOR AUSTENITIC STEEL 16Cr-12Ni-2Mo, TYPE 316



$$B := 6200 \text{ psi} \quad @ 400 \text{ F}$$

The maximum allowable working external pressure is then given by :

$$P_a := \frac{4B}{3 \left( \frac{2R_{i\_pv}}{t_{pv}} \right)} \quad P_a = 4.1 \text{ bar} \quad -P_{\min} = 1.5 \text{ bar}$$

$$P_a > -P_{\min} = 1$$

## Flange thickness:

inner radius                      max. allowable pressure  
 $R_{i\_pv} = 0.68 \text{ m}$                $P = 15.4 \text{ bar}$               (gauge pressure)

The flange design for O-ring sealing (or other self energizing gasket such as helicoflex) is "flat-faced", with "metal to metal contact outside the bolt circle". This design avoids the high flange bending stresses found in a raised face flange (of Appendix 2) and will result in less flange thickness. The rules for this design are found only in sec VIII division 1 under Appendix Y, and must be used with the allowable stresses of division 1. Flanges and shells will be fabricated from 316Ti (ASME spec SA-240) stainless steel plate. Plate samples will be helium leak checked before fabrication, as well as ultrasound inspected for flat laminar flaws which may create leak paths. The flange bolts and nuts will be inconel 718, (UNS N77180) as this is the highest strength non-corrosive material allowed for bolting.

We will design with enough flange strength to accomodate usin a Helicoflex 5mm gasket (smallest size possible) specially designed with a maximum sealing ofrce of 70 N/mm.

Maximum allowable material stresses, for sec VIII, division 1 rules from ASME 2010 Pressure Vessel code, sec. II part D, table 2A (division 1 only):

Maximum allowable design stress for flange

$$S_f := S_{\text{max\_316Ti\_div1}} \quad S_f = 137.9 \text{ MPa} \quad S_f = 2 \times 10^4 \text{ psi}$$

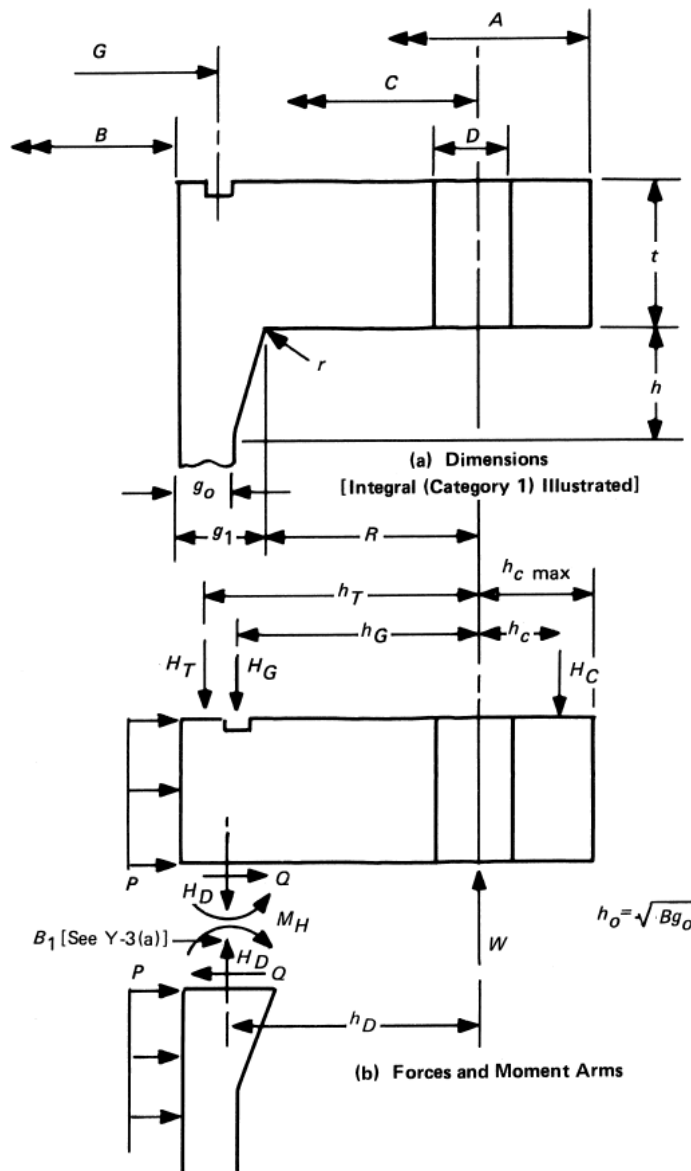
Maximum allowable design stress for bolts, from ASME 2010 Pressure Vessel code, sec. II part D, table 3

$$\text{Inconel 718 (UNS N07718)} \quad S_{\text{max\_N07718}} := 37000 \text{ psi}$$

$$S_b := S_{\text{max\_N07718}} \quad S_b = 255.1 \text{ MPa}$$

From sec. VIII div 1, non-mandatory appendix Y for bolted joints having metal-to-metal contact outside of bolt circle. First define, per Y-3:

FIG. Y-3.2 FLANGE DIMENSIONS AND FORCES



hub thickness at flange (no hub)

corner radius:

$$g_0 := t_{pv} \quad g_1 := t_{pv} \quad g_0 = 10 \text{ mm} \quad g_1 = 10 \text{ mm} \quad r_1 := \max(.25g_1, 5 \text{ mm}) \quad r_1 = 5 \text{ mm}$$

Flange OD

$$A := 1.47 \text{ m}$$

Flange ID

$$B := 2R_{i_{pv}} \quad B = 1.36 \text{ m}$$

define:

$$B_1 := B + g_1 \quad B_1 = 1.37 \text{ m}$$

Bolt circle (B.C.) dia, C:

$$C := 1.43 \cdot \text{m}$$

Gasket dia

$G := 2(R_{i\_pv} + .65\text{cm})$        $G = 1.373\text{m}$       O-ring mean radius as measured in CAD model       $68.65 \cdot 2 = 137.3$   
 Force of Pressure on head

$$H := .785G^2 \cdot \text{MAWP}_{pv} \quad H = 2.31 \times 10^6 \text{ N}$$

Sealing force, per unit length of circumference:

for O-ring, 0.275" dia., shore A 70  $F = \sim 5$  lbs/in for 20% compression, (Parker O-ring handbook); add 50% for smaller second O-ring. (Helicoflex gasket requires high compression, may damage soft Ti surfaces, may move under pressure unless tightly backed, not recommended)

Helicoflex has equivalent formulas using Y as the unit force term and gives several possible values.

for 5mm HN200 with aluminum jacket:

$$Y_1 := 120 \frac{\text{N}}{\text{mm}} \text{ min value for our pressure and required leak rate (He)} \quad Y_2 := 220 \frac{\text{N}}{\text{mm}} \text{ recommended value for large diameter seals, regardless of pressure or leak rate}$$

for gasket diameter       $D_j := G$        $D_j = 1.373\text{m}$

Force is then either of:

$$F_m := \pi D_j \cdot Y_1 \quad \text{or} \quad F_j := \pi D_j \cdot Y_2$$

$$F_m = 5.176 \times 10^5 \text{ N} \quad F_j = 9.489 \times 10^5 \text{ N}$$

Helicoflex recommends using Y2 ( 220 N/mm) for large diameter seals, even though for small diameter one can use the greater of Y1 or  $Y_m = (Y_2 \cdot (P/P_u))$ . For 15 bar Y1 is greater than Ym but far smaller than Y2. Sealing is less assured, but will be used in elastic range and so may be reusable. Flange thickness and bolt load increase quite substantially when using Y2 as design basis, which is a large penalty. We plan to recover any Xe leakage, as we have a second O-ring outside the first and a sniff port in between, so we thus design for Y1 (use  $F_m$ ) and "cross our fingers" : if it doesn't seal we use an O-ring instead and recover permeated Xe with a cold trap. Note: in the cold trap one will get water and N2, O2, that permeates through the outer O-ring as well.

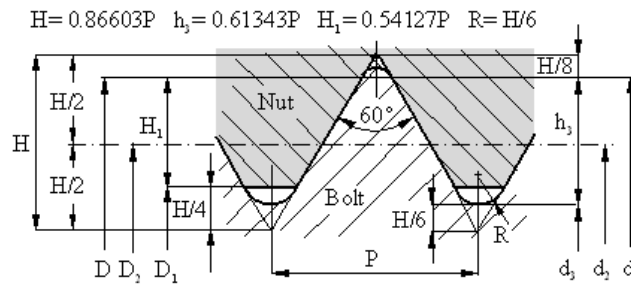
Start by making trial assumption for number of bolts, nominal bolt dia., pitch, and bolt hole dia D,

$$n := 132 \quad d_b := 16\text{mm} \quad \text{maximum number of bolts possible, using narrow washers:} \quad n_{\max} := \text{trunc}\left(\frac{\pi C}{2.0 d_b}\right) \quad n_{\max} = 140$$

Choosing ISO fine thread, to maximize root dia.; thread depth is:

$$p_t := 1.0\text{mm} \quad h_3 := .6134 p_t$$

using nomenclature and formulas from this chart at <http://www.tribology-abc.com/calculators/metric-iso.htm>



metric screw threads ISO 724 (DIN 13 T1)								
Nominal diameter d = D	Pitch P	root radius r	pitch diameter d2=D2	minor diameter d3	D1	thread height h3	H1	drill diameter mm
M 1.00	0.25	0.036	0.838	0.693	0.729	0.153	0.135	0.75
M 1.10	0.25	0.036	0.938	0.793	0.829	0.153	0.135	0.85
M 1.20	0.25	0.036	1.038	0.893	0.929	0.153	0.135	0.95
M 1.40	0.30	0.043	1.205	1.032	1.075	0.184	0.162	1.10
M 1.60	0.35	0.051	1.373	1.171	1.221	0.215	0.189	1.25
M 1.80	0.35	0.051	1.573	1.371	1.421	0.215	0.189	1.45
M 2.00	0.40	0.058	1.740	1.509	1.567	0.245	0.217	1.60
M 2.20	0.45	0.065	1.908	1.648	1.713	0.276	0.244	1.75
M 2.50	0.45	0.065	2.208	1.948	2.013	0.276	0.244	2.05
M 3.00	0.50	0.072	2.675	2.387	2.459	0.307	0.271	2.50
M 3.50	0.60	0.087	3.110	2.764	2.850	0.368	0.325	2.90
M 4.00	0.70	0.101	3.545	3.141	3.242	0.429	0.379	3.30
M 4.50	0.75	0.108	4.013	3.580	3.688	0.460	0.406	3.80
M 5.00	0.80	0.115	4.480	4.019	4.134	0.491	0.433	4.20
M 6.00	1.00	0.144	5.350	4.773	4.917	0.613	0.541	5.00
M 7.00	1.00	0.144	6.350	5.773	5.917	0.613	0.541	6.00
M 8.00	1.25	0.180	7.188	6.466	6.647	0.767	0.677	6.80
M 9.00	1.25	0.180	8.188	7.466	7.647	0.767	0.677	7.80
M 10.00	1.50	0.217	9.026	8.160	8.376	0.920	0.812	8.50
M 11.00	1.50	0.217	10.026	9.160	9.376	0.920	0.812	9.50
M 12.00	1.75	0.253	10.863	9.853	10.106	1.074	0.947	10.20
M 14.00	2.00	0.289	12.701	11.546	11.835	1.227	1.083	12.00
M 16.00	2.00	0.289	14.701	13.546	13.835	1.227	1.083	14.00
M 18.00	2.50	0.361	16.376	14.933	15.394	1.534	1.353	15.50
M 20.00	2.50	0.361	18.376	16.933	17.294	1.534	1.353	17.50

<--- use h3 for 1.0 mm pitch

<--- use H1 for 1.5mm pitch

Bolt root dia. is then:

$$d_3 := d_b - 2h_3 \quad d_3 = 14.7732 \text{ mm}$$

Total bolt cross sectional area:

$$A_b := n \cdot \frac{\pi}{4} d_3^2 \quad A_b = 226.263 \text{ cm}^2$$

Check bolt to bolt clearance, here we use narrow thick washers (28mm OD) under the 24mm nuts (28mm is also corner to corner distance on nut), we adopt a minimum bolt spacing of 2x the nominal bolt diameter (to give room for a 24mm socket) :

$$\pi C - 2.0n \cdot d_b \geq 0 = 1 \quad \text{actual bolt to bolt distance: } \frac{\pi C}{n} = 34.034 \text{ mm}$$

Check nut, washer, socket clearance:  $OD_w := 2d_b$

this is for standard narrow washers, and for wrench sockets which more than cover the nut width across corners

$$0.5C - (0.5B + g_1 + r_1) \geq 0.5OD_w = 1$$

Flange hole diameter, minimum for clearance :

$$D_{\text{tmin}} := d_b + 2\text{mm} \quad D_{\text{tmin}} = 18 \text{ mm}$$

We will thread some of these clearance holes for M20-1.5 bolts to allow the head retraction fixture to be bolted up the the flange. The effective diameter of these holes will be the average of nominal and minimum diameters. These holes must contain thin (0.5mm max thk.) metal sleeves when not in use to avoid thread interference with flange bolts. The actual clearance holes will be 18mm.

$$H_1 := .812\text{mm} \quad \text{http://www.tribology-abc.com/calculators/metric-iso.htm}$$

$$d_{\min\_20\_1.5} := 20\text{mm} - 2 \cdot H_1 \quad d_{\min\_20\_1.5} = 1.838\text{cm} \quad d_{\min\_20\_1.5} \geq D_{\min} = 1$$

$$D_e := 0.5(20\text{mm} + d_{\min\_20\_1.5}) \quad D_e = 1.919\text{cm}$$

Set:

$$D_t := D_e$$

$$D_t > D_{\min} = 1$$

Compute Forces on flange:

We use a unit gasket seating force of Y1 above

$$H_G := F_m \quad H_G = 5.176 \times 10^5 \text{ N}$$

$$h_G := 0.5(C - G) \quad h_G = 2.85\text{cm}$$

$$H_D := .785 \cdot B^2 \cdot P \quad H_D = 2.266 \times 10^6 \text{ N}$$

$$h_D := D_t \quad h_D = 1.919\text{cm}$$

$$H_T := H - H_D \quad H_T = 4.353 \times 10^4 \text{ N}$$

$$h_T := 0.5(C - B) \quad h_T = 35\text{mm}$$

Total Moment on Flange

$$M_P := H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad M_P = 5.976 \times 10^4 \text{ J}$$

## Appendix Y Calc

$$P = 15.4\text{bar}$$

Choose values for plate thickness and bolt hole dia:

$$t := 3.6\text{cm} \quad D := D_t \quad D = 1.919\text{cm}$$

Going back to main analysis, compute the following quantities:

$$\beta := \frac{C + B_1}{2B_1} \quad \beta = 1.022 \quad h_C := 0.5(A - C) \quad h_C = 0.02\text{m}$$

$$a := \frac{A + C}{2B_1} \quad a = 1.058 \quad AR := \frac{n \cdot D}{\pi \cdot C} \quad AR = 0.564 \quad h_0 := \sqrt{B \cdot g_0}$$

$$r_B := \frac{1}{n} \left( \frac{4}{\sqrt{1 - AR^2}} \operatorname{atan} \left( \sqrt{\frac{1 + AR}{1 - AR}} \right) - \pi - 2AR \right) \quad r_B = 7.462 \times 10^{-3} \quad h_0 = 0.117\text{m}$$

We need factors F and G, most easily found in figs 2-7.2 and 7.3 (Appendix 2)

$$\text{since } \frac{g_1}{g_0} = 1 \quad \text{these values converge to} \quad F := 0.90892 \quad V := 0.550103$$

## Y-5 Classification and Categorization

We have identical (class 1 assembly) integral (category 1) flanges, so from table Y-6.1, our applicable equations are (5a), (7)-(13), (14a), (15a), (16a)

$$J_S := \frac{1}{B_1} \left( \frac{2 \cdot h_D}{\beta} + \frac{h_C}{a} \right) + \pi r_B \quad J_S = 0.065 \quad J_P := \frac{1}{B_1} \left( \frac{h_D}{\beta} + \frac{h_C}{a} \right) + \pi r_B \quad J_P = 0.051$$

$$(5a) \quad F' := \frac{\xi_0^2 (h_0 + F \cdot t)}{V} \quad F' = 2.715 \times 10^{-5} \text{ m}^3 \quad M_P = 5.976 \times 10^4 \text{ N} \cdot \text{m}$$

$$A = 1.47 \text{ m} \quad B = 1.36 \text{ m}$$

$$K := \frac{A}{B} \quad K = 1.081 \quad Z := \frac{K^2 + 1}{K^2 - 1} \quad Z = 12.883$$

$$f := 1$$

$$t_s := 0 \text{ mm} \quad \text{no spacer between flanges}$$

$$l := 2t + t_s + 0.5d_b \quad l = 8 \text{ cm}$$

sec Y-6.2(a)(3)

Elastic constants

<http://www.hightempmetals.com/tech-data/hitempInconel718data.php>

$$(7-13) \quad E := E_{SS\_aus} \quad E_{Inconel\_718} := 208 \text{ GPa} \quad E_{bolt} := E_{Inconel\_718}$$

$$M_S := \frac{-J_P \cdot F' \cdot M_P}{t^3 + J_S \cdot F'} \quad M_S = -1.7 \times 10^3 \text{ N} \cdot \text{m}$$

$$\theta_B := \frac{5.46}{E \cdot \pi t^3} (J_S \cdot M_S + J_P \cdot M_P) \quad \theta_B = 5.663 \times 10^{-4} \quad E \cdot \theta_B = 109.291 \text{ MPa}$$

$$H_C := \frac{M_P + M_S}{h_C} \quad H_C = 2.903 \times 10^6 \text{ N}$$

$$W_{m1} := H + H_G + H_C \quad W_{m1} = 5.73 \times 10^6 \text{ N}$$

### Compute Flange and Bolt Stresses

$$\sigma_b := \frac{W_{m1}}{A_b} \quad \sigma_b = 253.2 \text{ MPa} \quad S_b = 255.1 \text{ MPa}$$

$$r_E := \frac{E}{E_{bolt}} \quad r_E = 0.928$$

$$S_i := \sigma_b - \frac{1.159 \cdot h_C^2 \cdot (M_P + M_S)}{a \cdot t^3 \cdot r_E \cdot B_1} \quad S_i = 247.9 \text{ MPa}$$

$$S_{R\_BC} := \frac{6(M_P + M_S)}{t^2 (\pi \cdot C - n \cdot D)} \quad S_{R\_BC} = 137.1 \text{ MPa} \quad S_f = 137.9 \text{ MPa}$$

$$S_{R\_ID1} := - \left( \frac{2F \cdot t}{h_0 + F \cdot t} + 6 \right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_{R\_ID1} = 1.97 \text{ MPa}$$

$$S_{T1} := \frac{t \cdot E \cdot \theta_B}{B_1} + \left( \frac{2F \cdot t \cdot Z}{h_0 + F \cdot t} - 1.8 \right) \cdot \frac{M_S}{\pi B_1 \cdot t^2} \quad S_{T1} = 1.69 \text{ MPa}$$

$$S_{T3} := \frac{t \cdot E \cdot \theta_B}{B_1} \quad S_{T3} = 2.872 \text{ MPa}$$

$$S_H := \frac{h_0 \cdot E \cdot \theta_B \cdot f}{0.91 \left( \frac{g_1}{g_0} \right)^2 B_1 \cdot V} \quad S_H = 18.584 \text{ MPa}$$

**Y-7 Flange stress allowables:**  $S_f = 137.9 \text{ MPa}$

- (a)  $\sigma_b < S_b = 1$
- (b) (1)  $S_H < 1.5S_f = 1$   $S_n$  not applicable
- (b) (2) not applicable
- (c)  $S_{R\_BC} < S_f = 1$   
 $S_{R\_ID1} < S_f = 1$
- (d)  $S_{T1} < S_f = 1$   
 $S_{T3} < S_f = 1$
- (e)  $\frac{S_H + S_{R\_BC}}{2} < S_f = 1$   
 $\frac{S_H + S_{R\_ID1}}{2} < S_f = 1$
- (f) not applicable

Bolt force

$$F_{\text{bolt}} := \sigma_b \cdot .785 \cdot d_b^2 \quad F_{\text{bolt}} = 1.144 \times 10^4 \text{ lbf}$$

Bolt torque required, minimum:

$$T_{\text{bolt\_min}} := 0.2F_{\text{bolt}} \cdot d_b \quad T_{\text{bolt\_min}} = 162.9 \text{ N} \cdot \text{m} \quad T_{\text{bolt\_min}} = 120.1 \text{ lbf} \cdot \text{ft} \quad \text{for pressure test use 1.5x this value}$$

This is the minimum amount of bolt preload needed to assure joint does not open under pressure. and additional amount for bolt preload is needed to maintain a minimum shear resistance to assure head does not slide downward from weight. Non-mandatory Appendix S of div. 1 makes permissible higher bolt stresses than indicated above when needed to assure full gasket sealing and other conditions. This is consistent with proper preloaded joint practice, in which, for properly designed joints where connection stiffness is much greater than bolt stiffness,

Additional Calculations:

Shear stress in inner flange lip from shield (could happen only if flange bolts come loose, are left loose, or if joint

opens under pressure, otherwise friction of faces will support shield, given additional tension, as permissible under non-mandatory Appendix S above )

masses of Copper shielding in cyl and heads

$$t_{Cu} = 0.12 \text{ m} \quad \rho_{Cu} = 9 \times 10^3 \frac{\text{kg}}{\text{m}^3}$$

$$M_{sh\_head} := \rho_{Cu} \cdot \pi R_{i\_pv}^2 \cdot t_{Cu} \quad M_{sh\_head} = 1.569 \times 10^3 \text{ kg}$$

$$M_{sh\_cyl} := \rho_{Cu} \cdot 2\pi \cdot R_{i\_pv} \cdot t_{Cu} \cdot L_{ff} \quad M_{sh\_cyl} = 7.383 \times 10^3 \text{ kg}$$

$$t_{lip} := 3 \text{ mm}$$

Shear stress in lip:

$$\tau_{lip} := \frac{M_{sh\_head} \cdot g}{R_{i\_pv} \cdot t_{lip}} \quad \tau_{lip} = 7.542 \text{ MPa}$$

Shear stress on O-ring land (section between inner and outer O-ring), from pressurized O-ring. This is assumed to be the primary stress. There is some edge moment but the "beam" is a very short one. This shear stress is not in the same direction as the nominal tangential (hoop) stress of the flange.

$$t_{land\_radial} := .36 \text{ cm} \quad w_{land\_axial} := .41 \text{ cm}$$

$$F_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot w_{land\_axial} \cdot P$$

$$A_{O\_ring\_land} := 2\pi R_{i\_pv} \cdot t_{land\_radial}$$

$$\tau_{land} := \frac{F_{O\_ring\_land}}{A_{O\_ring\_land}} \quad \tau_{land} = 1.778 \text{ MPa}$$

### Support Design using rules of div 2, part 4.15:

From the diagram below the rules are only applicable to flange attached heads if there is a flat cover or tubesheet inside, effectively maintaining the flanges circular. Since the PMT carrier plate and shielding is firmly bolted in, it serves this purpose and we may proceed. We must also compute the case with the heads attached, as there will be additional load

a) Design Method- although not specifically stated, the formulas for bending moments at the center and at the supports are likely based on a uniform loading of the vessel wall from the vessel contents. In this design, the internal weight (primarily of the copper shield) is applied at the flanges; there is no contact with the vessel shell. We calculate both ways and take the worst case.

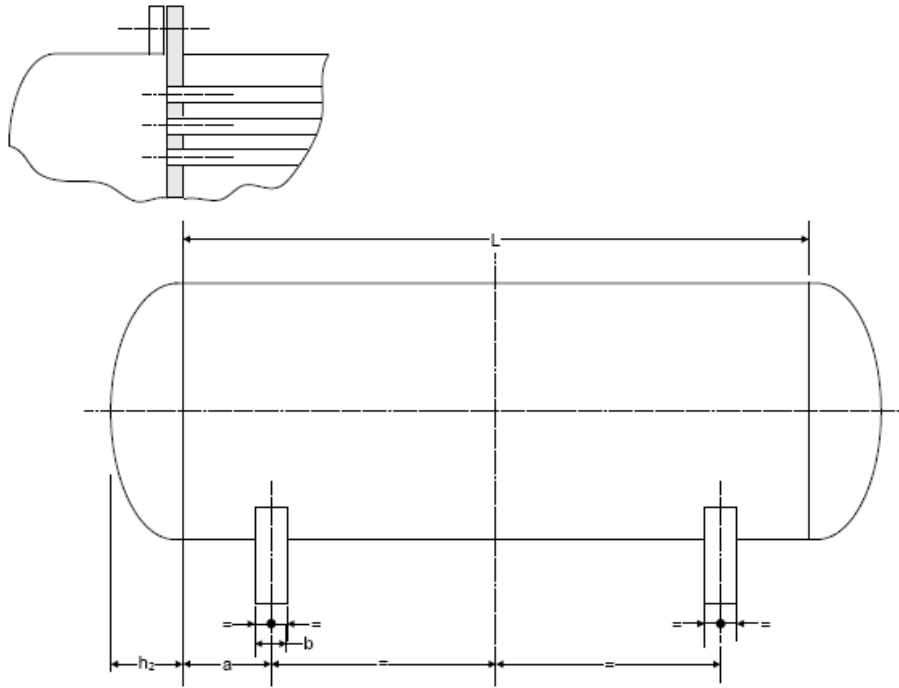


Figure 4.15.1 – Horizontal Vessel on Saddle Supports

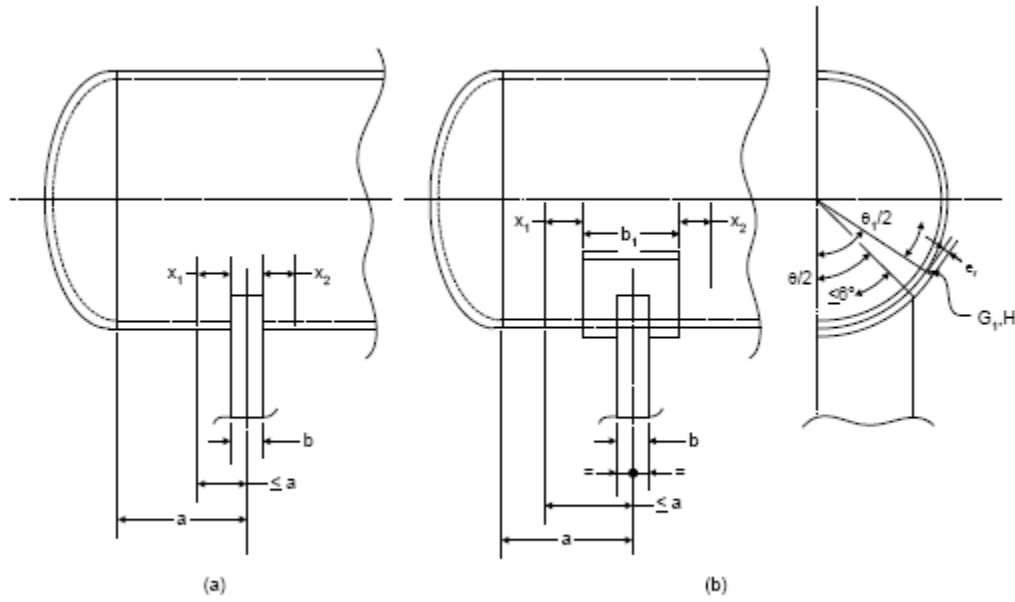


Figure 4.15.2 – Cylindrical Shell Without Stiffening Rings

$$L := L_{ff} \quad M_{tot} := 12000 \text{ kg}$$

$$b := 1.5 \text{ cm} \quad a := .18 L_{ff} \quad a = 28.8 \text{ cm} \quad \theta := 120 \text{ deg}$$

$$R_m := R_{i\_pv} + 0.5 t_{pv}$$

$$b_1 := \min \left[ \left( b + 1.56 \cdot \sqrt{R_m \cdot t_{pv}} \right), 2 \cdot a \right]$$

$$b_1 = 14.411 \text{ cm}$$

$$h_2 := 20 \text{ cm} \quad k := 0.1$$

$$\theta_1 := \theta + \frac{\theta}{12}$$

$$\theta_1 = 130 \text{ deg}$$

maximum reaction load at each support:

$$Q := 0.5 M_{tot} \cdot g \quad Q = 5.884 \times 10^4 \text{ N}$$

$$M_1 := -Q \cdot a \cdot \left( 1 - \frac{1 - \frac{a}{L} + \frac{R_m^2 - h_2^2}{2 \cdot a \cdot L}}{1 + \frac{4 h_2}{3 L}} \right)$$

$$M_1 = 1.73 \times 10^3 \text{ N} \cdot \text{m}$$

$$Q \cdot a = 1.695 \times 10^4 \text{ J}$$

$$M_2 := \frac{Q \cdot L}{4} \cdot \left[ \frac{1 + \frac{2 \cdot (R_m^2 - h_2^2)}{L^2}}{1 + \frac{4 \cdot h_2}{3 L}} - \frac{4 a}{L} \right]$$

$$M_2 = 9.993 \times 10^3 \text{ N} \cdot \text{m}$$

$$M_{1'} := Q \cdot a \quad M_{1'} = 1.695 \times 10^4 \text{ N} \cdot \text{m}$$

$$M_{2'} := M_{1'} \quad M_{2'} = 1.695 \times 10^4 \text{ N} \cdot \text{m}$$

$$T := \frac{Q(L - 2a)}{L + \frac{4h_2}{3}}$$

$$T = 3.228 \times 10^4 \text{ N}$$

#### 4.15.3.3 - longitudinal . stresses

distributed load (ASME assumption)

end load ( actual)

$$\sigma_1 := \frac{P \cdot R_m}{2t_{pv}} - \frac{M_2}{\pi R_m^2 t_{pv}} \quad \sigma_1 = 52.781 \text{ MPa}$$

$$\sigma_{1'} := \frac{P \cdot R_m}{2t_{pv}} - \frac{M_{2'}}{\pi R_m^2 t_{pv}} \quad \sigma_{1'} = 52.309 \text{ MPa}$$

$$\sigma_2 := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_2}{\pi R_m^2 t_{pv}} \quad \sigma_2 = 54.136 \text{ MPa}$$

$$\sigma_{2'} := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_{2'}}{\pi R_m^2 t_{pv}} \quad \sigma_{2'} = 54.608 \text{ MPa}$$

same stress at supports, since these are stiffened, as  $a < 0.5R_m$  and close to a torispheric head

$$a < 0.5R_m = 1$$

$$\sigma_3 := \frac{P \cdot R_m}{2t_{pv}} - \frac{M_1}{\pi R_m^2 t_{pv}} \quad \sigma_3 = 53.341 \text{ MPa}$$

$$\sigma_{3'} := \frac{P \cdot R_m}{2t_{pv}} - \frac{M_{1'}}{\pi R_m^2 t_{pv}} \quad \sigma_{3'} = 52.309 \text{ MPa}$$

$$\sigma_4 := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_1}{\pi R_m^2 t_{pv}} \quad \sigma_4 = 53.576 \text{ MPa}$$

$$\sigma_{4'} := \frac{P \cdot R_m}{2t_{pv}} + \frac{M_{1'}}{\pi R_m^2 t_{pv}} \quad \sigma_{4'} = 54.608 \text{ MPa}$$

#### 4.15.3.4 - Shear stresses

$$\Delta := \frac{\pi}{6} + \frac{5\theta}{12} \quad \Delta = 1.396$$

$$\alpha := 0.95 \left( \pi - \frac{\theta}{2} \right) \quad \alpha = 1.99$$

$$K_2 := \frac{\sin(\alpha)}{\pi - \alpha + \sin(\alpha) \cos(\alpha)} \quad K_2 = 1.171$$

here we use c), formula for cyl. shell with no stiffening rings and which is not stiffened by a formed head, flat cover or tubesheet. This is worst case, as we have a flange, which can be considered as one half of a stiffening ring pair for each support.

$$c) \quad \tau_1 := \frac{K_2 \cdot T}{\pi R_m \cdot t_{pv}} \quad \tau_1 = 1.756 \text{ MPa} \quad (4.15.14)$$

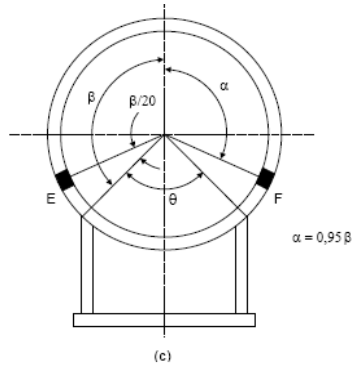
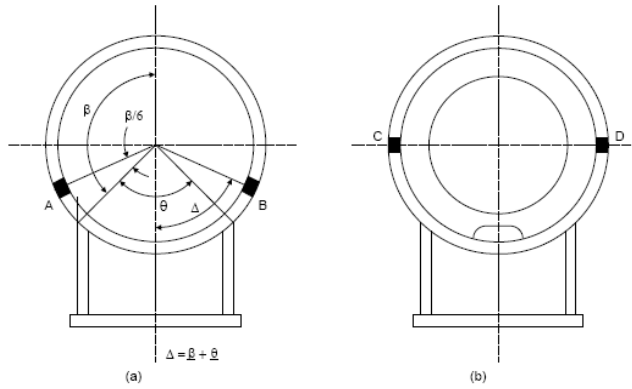


Figure 4.15.5 – Locations of Maximum Longitudinal Normal Stress and Shear Stress in the Cylinder



#### 4.15.3.5 Circumferential Stress

$$K_5 := \frac{1 + \cos(\alpha)}{\pi - \alpha + \sin(\alpha) \cdot \cos(\alpha)} \quad K_5 = 0.76$$

$$\beta := \pi - \frac{\theta}{2} \quad \beta = 2.094$$

$$K_6 := \frac{\frac{3 \cdot \cos(\beta)}{4} \cdot \left( \frac{\sin(\beta)}{\beta} \right)^2 - \frac{5 \cdot \sin(\beta) \cdot \cos(\beta)}{4 \cdot \beta} + \frac{\cos(\beta)^3}{2} - \frac{\sin(\beta)}{4 \cdot \beta} + \frac{\cos(\beta)}{4} - \beta \cdot \sin(\beta) \cdot \left[ \left( \frac{\sin(\beta)}{\beta} \right)^2 - \frac{1}{2} - \frac{\sin(2 \cdot \beta)}{4 \cdot \beta} \right]}{2 \cdot \pi \cdot \left[ \left( \frac{\sin(\beta)}{\beta} \right)^2 - \frac{1}{2} - \frac{\sin(2 \cdot \beta)}{4 \cdot \beta} \right]}$$

$$K_6 = -0.221$$

$$\frac{a}{R_m} < 0.5 = 1$$

$$K_7 := \frac{K_6}{4} \quad K_7 = -0.055$$

a) Max circ bending moment

1) Cyl shell without a stiffening ring

$$M_{\beta} := K_7 \cdot Q \cdot R_m \quad M_{\beta} = -2.223 \times 10^3 \text{ N}\cdot\text{m}$$

c) Circ. stress in shell, without stiffening rings

$$x_1 := 0.78 \sqrt{R_m \cdot t_{pv}} \quad x_1 = 6.456 \text{ cm} \quad x_2 := x_1 \quad k = 0.1$$

$$\sigma_6 := \frac{-K_5 \cdot Q \cdot k}{t_{pv} \cdot (b + x_1 + x_2)} \quad \sigma_6 = -3.104 \text{ MPa}$$

$$L < 8R_m = 1$$

$$L = 1.6 \text{ m}$$

$$b_1 = 14.411 \text{ cm}$$

$$\sigma_7 := \frac{-Q}{4t_{pv} \cdot (b + x_1 + x_2)} - \frac{12K_7 \cdot Q \cdot R_m}{L \cdot t_{pv}^2} \quad \sigma_7 = 156.484 \text{ MPa} \quad (4.15.25)$$

too high; we need a reinforcement plate of thickness;

$$t_r := 0.5t_{pv} \quad \text{strength ratio: } \eta := 1 \quad (4.15.29)$$

$$\sigma_{7r} := \frac{-Q}{4(t_{pv} + \eta \cdot t_r) \cdot b_1} - \frac{12K_7 \cdot Q \cdot R_m}{L \cdot (t_{pv} + \eta \cdot t_r)^2} \quad \sigma_{7r} = 67.28 \text{ MPa} \quad (4.15.28)$$

3) f) Acceptance Criteria

$$S = 1.379 \times 10^8 \text{ Pa} \quad S = 2 \times 10^4 \text{ psi}$$

$$|\sigma_{7r}| < 1.25S = 1$$

4) this section not applicable as  $t_r > 2t_{pv} = 0$

4.15.3.6 - Saddle support, horizontal force given below must be resisted by low point of saddle ( where height =  $h_s$ )

$$F_h := Q \cdot \left( \frac{1 + \cos(\beta) - 0.5 \cdot \sin(\beta)^2}{\pi - \beta + \beta \cdot \sin(\beta) \cos(\beta)} \right) \quad F_h = 5.242 \times 10^4 \text{ N} \quad h_s := 9 \text{ cm}$$

$$\sigma_h := \frac{F_h}{b \cdot h_s} \quad \sigma_h = 38.833 \text{ MPa}$$